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THE IMPORTANCE OF AERODYNAMICS IN THE DESIGN OF INTRA-URBAN TRAINS TRAVELING IN TUNNELS

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and
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Aerodynamics can be a major factor in the design and operation of intraurban subway-train transportation systems. In order to develop an adequate understanding of the aerodynamic characteristics of such systems, an experimental program was carried out. The major portion of the testing was conducted under equilibrium, incompressible conditions so that the fundamental aerodynamic characteristics could be isolated. The effects of geometric parameters (such as train speed, blockage ratio, wall roughness, and train and tunnel length) upon train drag and tunnel flow velocities were determined and compared with a simple theoretical model. The effect of aerodynamic forces upon typical subway-train operations is shown in order to give proper perspective to the importance of aerodynamics.

INTRODUCTION

Many domestic transportation systems are currently being studied and built which require vehicles to operate in tunnels under high blockage conditions. As a consequence, aerodynamic forces may be one to two orders of magnitude greater than if the train were operating outside the tunnel.* Power requirements due to aerodynamic drag, heat rejection into the tunnel from the train power source, and passenger discomfort due to air blast at the stations contribute problems to an already complex system. It is necessary to quantitatively predict the aerodynamic drag, resultant air flow velocities and pressure pulses to optimize the design of such systems. It has been predicted that many tens of billions of dollars will be spent in the next 10 to 15 years on new subway rapid transit systems and the modernization and expansion of existing ones. It is therefore incumbent upon the designers and planners to have a soundly based appreciation of all the parameters involved.

The concepts and examples which are presented in this article presume that a vehicle is traveling under atmospheric incompressible conditions. The same principles apply in partially evacuated tunnel applications (Reference 1) and, naturally, aerodynamic factors are non-existent in a hard vacuum environment (Reference 2). Compressibility effects on the aerodynamic characteristics of a system are a function of both the tunnel pressure and train velocity (Reference

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*San Francisco's BART System incorporates some 28 miles of tunnels; the Washington, D.C. rapid transit system has 47 route miles of subway and many more miles of tunnels are in various stages of planning in Southern California, Pittsburg, Atlanta, Baltimore, and elsewhere.

3) and must be considered for proposed high speed inter-urban travel where speeds may be far in excess of 100 mph.

The purpose of this paper is to indicate the importance of some of the aerodynamic aspects of intra-urban subways as a guide to an appreciation of various parameters concerning the operation of such systems.

Considerable experimental work (References 4 through 11) has been done previously using both small scale and actual operating systems in order to determine the aerodynamic characteristics of trains traveling through tunnels. None of these studies, however, made direct aerodynamic drag measurements under the controlled, ideal conditions of steady-state incompressible-flow at full-scale Reynolds numbers. Therefore an extensive experimental program was carried out at JPL in which the effects of many system variables were studied (train blockage and length, tunnel length, train and tunnel roughness, and Reynolds number)* Many hundreds of runs were taken, reduced and analyzed from the VICS-120 facility (Reference 12). Only a few representative runs are presented in this paper to demonstrate the effect of a tunnel on the aerodynamic characteristics of a train. A complete program and data presentation has been published in Reference 13.

BASIC STEADY-STATE AERODYNAMIC CHARACTERISTICS

Power Requirements

There are many parameters which affect the aerodynamic characteristics of vehicles traveling in tunnels. Some of the more important ones will be discussed for a system operating under steady-state conditions where only aerodynamic and rolling friction forces must be overcome by the propulsion units. Experimental data will be presented which were obtained in the VICS-120 facility with a simplified theoretical model used as a guide for fairing the data (see Appendix and Reference 14).

The vehicle blockage ratio, σ, is probably the single most important parameter effecting the drag of trains traveling in tunnels. The horsepower required to overcome this drag is presented in Figure 1. A blockage ratio of zero corresponds to a train outside the tunnel. Notice that a blockage ratio of .75, the horsepower necessary to cruise at 70 miles per hour in a two mile unvented tunnel is an order of magnitude greater than that had the train been outside the tunnel. This example is for the most favorable case where both the train and tunnel walls are aerodynamically smooth. Train detailing, such as windows and doors, as well as wheel bogeys can account for another 50% increase in drag and power requirements at a blockage ratio near 0.6 as shown.

Since vehicle length is an important geometric parameter which can change daily (or even hourly as passenger loads require) a clear understanding of its influence on power requirements is necessary. In operation outside of tunnels adding cars adds essentially equal increments of skin friction drag. The con-

* This experimental work was jointly sponsored by the Department of Transportation's Urban Mass Transportation Administration and the Transit Development Corporation, Inc. It is part of a project entitled "Ventilation and Environmental Control in Subway Rapid Transit Systems," the end result of which will be a Subway Environmental Design Handbook.

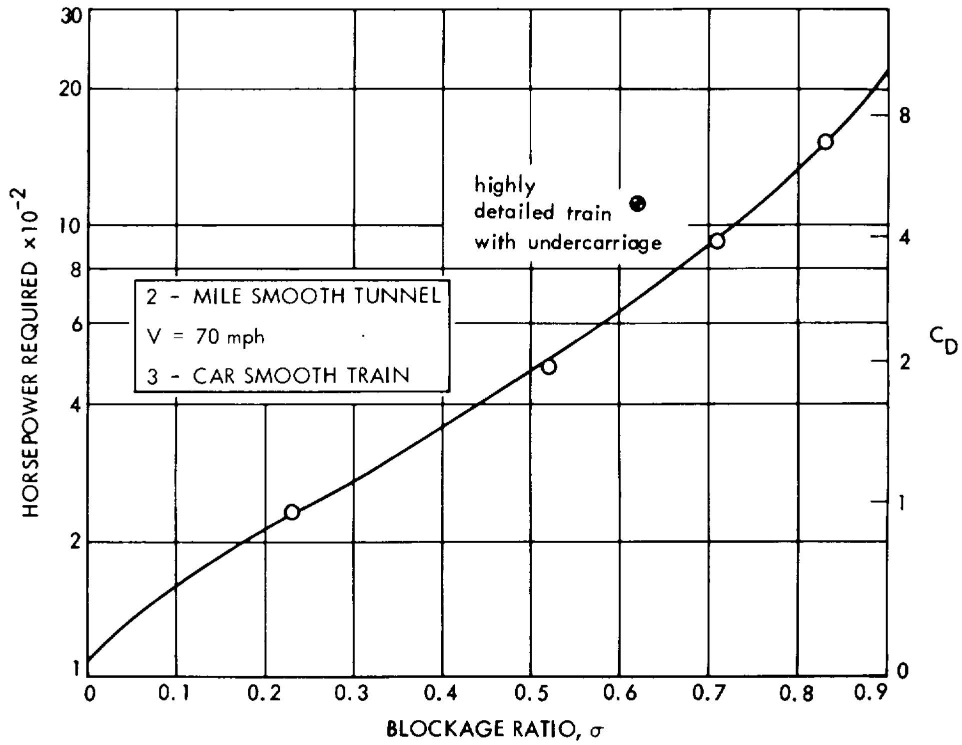


FIGURE 1. Horsepower Required to Overcome Aerodynamic Drag as a Function of Blockage Ratio

straint of the tunnel wall changes this to some extent. The horsepower necessary to overcome additional car drag is shown in Figure 2. Only one blockage ratio is presented, since the trend is not sensitive to blockage ratio except at very low blockage ratios (approaching the free air case). As the blockage ratio approaches unity, the number of cars theoretically has a vanishing effect on drag. It should be noted that in modern subway systems each car usually has its own propulsion unit; hence, the power available increases linearly while the power required to overcome aerodynamic drag does not. Increasing train length, then, may be a simple and convenient method of improving the power characteristics of the system.

As the tunnel length is increased the power required to operate also increases. Figure 3 shows that the form of the penalty is a strong function of the blockage ratio. If the tunnel length is increased from two miles to ten miles, the cruise power requirement for the 71% blockage train may nearly double; however, only about 20% more power is necessary for a 23% blockage system. It is clear, then, that any method of decreasing the tunnel length, or effective length, is beneficial, especially at higher blockage ratios. Properly designed vent shafts and elimination of internal ribbing effectively decrease the tunnel length.

Reynolds number is the most important scaling parameter in low speed aerodynamics. It is an indication of whether or not the flow is behaving as it would

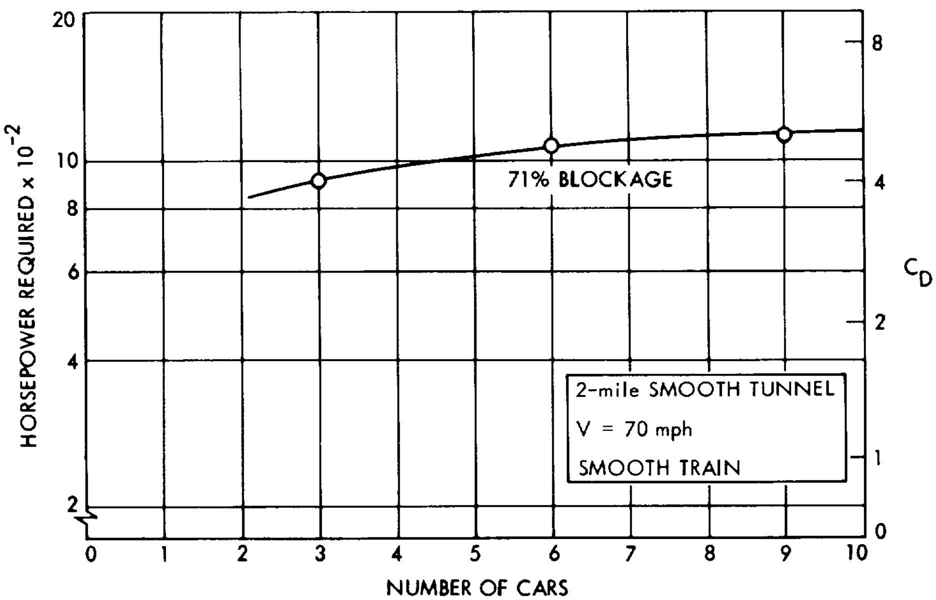


FIGURE 2. Horsepower Required to Overcome Aerodynamic Drag as a Function of Train Length

in the full-scale case. The drag coefficient varies with Reynolds number in much the same fashion as the smooth-wall friction factor. Figure 4 presents this coefficient for a 71% blockage vehicle under two conditions: smooth and ribbed tunnel walls (consisting of six-inch high bulkheads on two-and-a-half foot centers). At full-scale Reynolds numbers the drag (or cruise power requirements) may be three times as high in a ribbed tunnel length as in a smooth tunnel. As tunnel length is increased to infinity the relative penalty for the ribbed tunnel is decreased since the flow velocity is already reduced.

Tunnel Flow Velocity

The major difference between a train operating in free air or inside a tunnel is that in the case of the latter, the control volume has solid walls. That is, the vehicle must contend with a specific volume (hence, mass) of air, since the fluid is constrained to flow within definite boundaries. As the train moves through the tunnel it must move (or attempt to move) this column of air at some proportion of the train's own velocity. Figure 5 shows the equilibrium flow velocity caused by a train at 70 mph as a function of the blockage ratio. If the vehicle had a blockage ratio of unity, the steady state incompressible flow in an unvented tunnel would be moving at exactly the vehicle speed (unless the tunnel exit were sealed). Likewise, if the blockage ratio were zero, which is the case in free air, the tunnel flow velocity would be zero.

The tunnel flow velocity is affected to a lesser degree by the number of cars making up the train for much the same reasons noted when discussing power

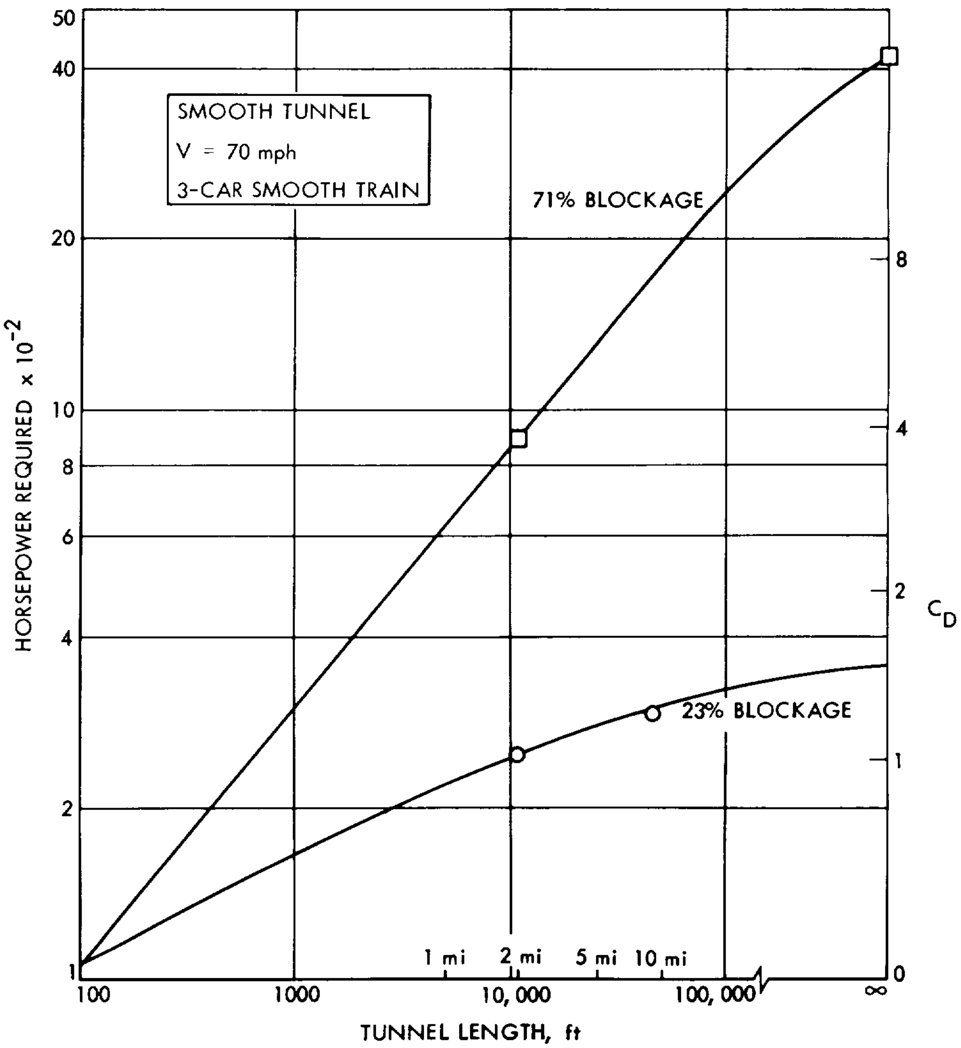


FIGURE 3. Horsepower Required to Overcome Aerodynamic Drag as a Function of Tunnel Length

requirements. Figure 6 is presented for only one blockage ratio as the asymptotic trend is the same but delayed a few cars at lower blockage ratios.

The speed at which a vehicle can cause the fluid in the tunnel to travel is governed by the vehicle drag and the fluid's resistance to flow. This resistance is a function of the friction force along the tunnel wall. If the tunnel is infinitely long, the fluid's friction force will also be infinite and no amount of power will set it into motion if it is incompressible. As the tunnel length is decreased, the fluid moves more easily and the flow velocity increases. Figure 7 presents the situation for two blockage ratios. Methods to decrease the effective tunnel length, such as venting, would have significant effects on both the tunnel flow velocity

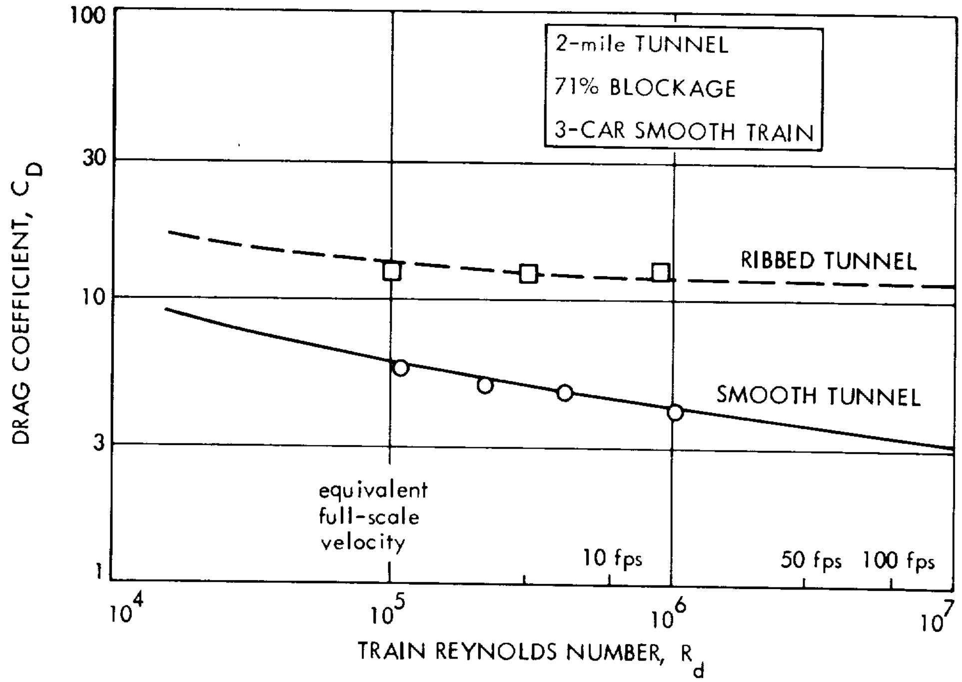


FIGURE 4. Drag Coefficient Variation with Reynolds Number

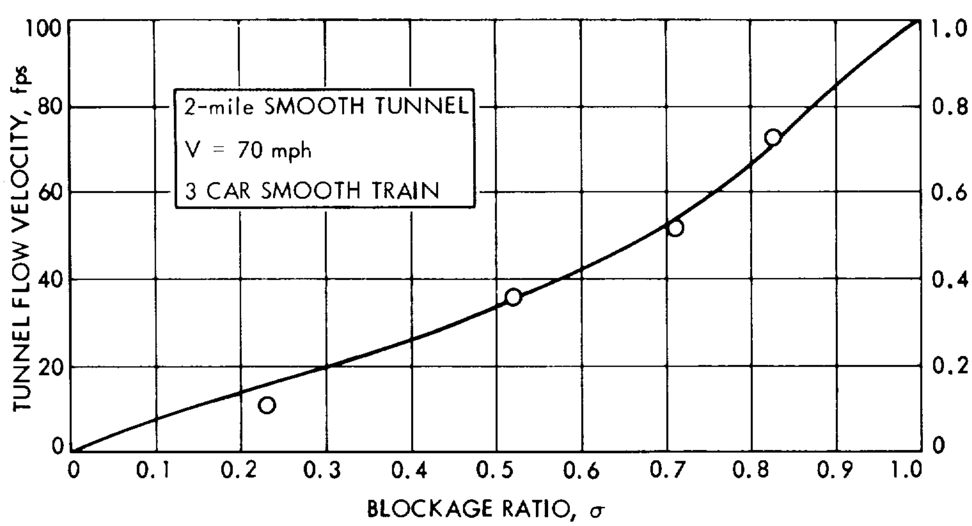


FIGURE 5. Air Flow Velocity in the Tunnel as a Function of Blockage Ratio

and the vehicle drag. On the other hand, the effective tunnel length is increased by adding ribs to the inner tunnel wall, hence decreasing the flow velocity. Figure 8 presents the tunnel flow velocity as a function of Reynolds number for both the smooth and ribbed tunnel. At cruise speed, the flow velocity in the ribbed tunnel is just about half that in the smooth tunnel.

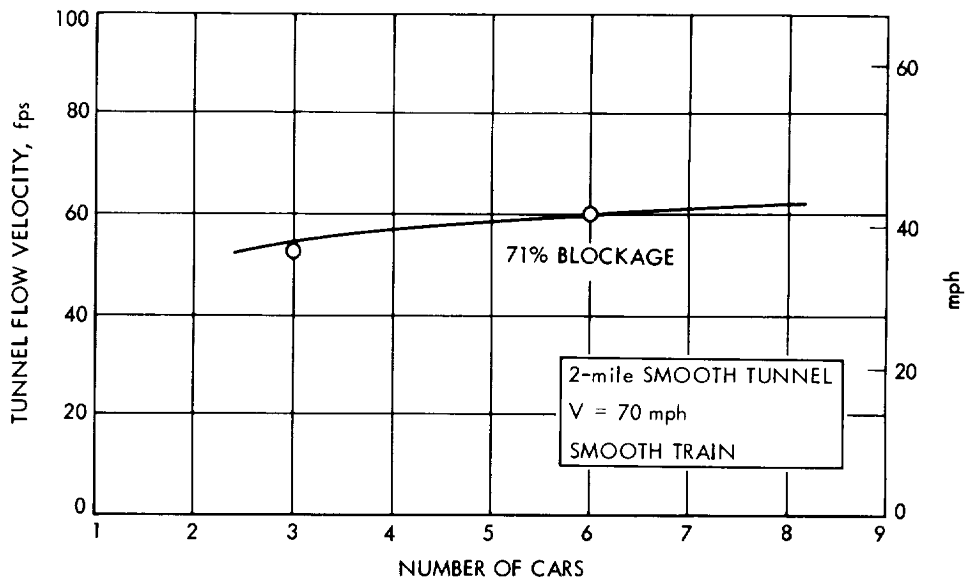


FIGURE 6. Air Flow Velocity in the Tunnel as a Function of Train Length

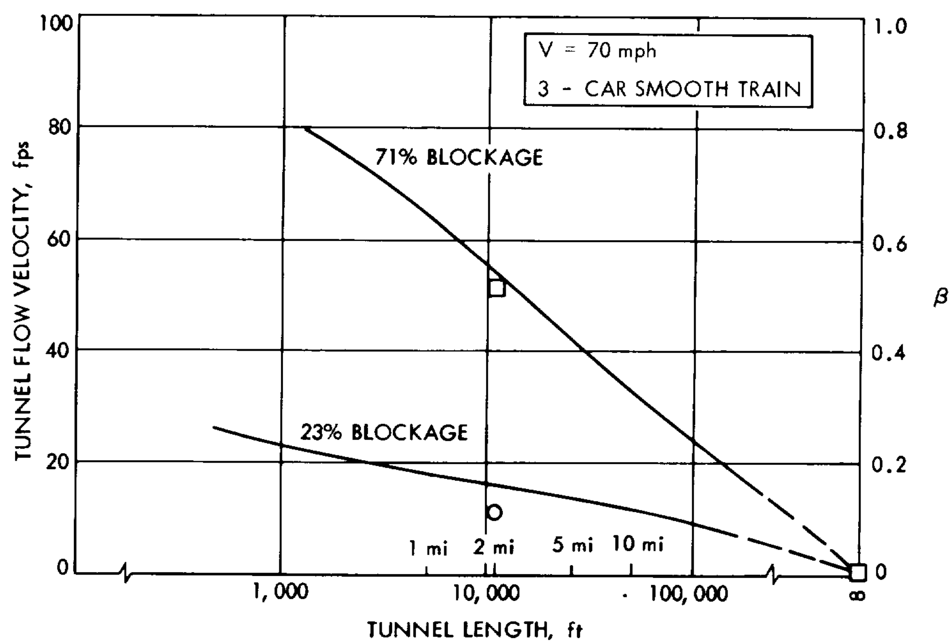


FIGURE 7. Air Flow Velocity in Tunnel as a Function of Tunnel Length

The pressure rise in a station due to a train entering a tunnel a mile or so upstream and the resulting passenger discomfort are major concerns. The pressure gradient has been measured to be as large as 1½ psi/sec in both the station and inside the lead car on systems now in operation. Tunnel venting alleviates the situation to some degree by reducing the magnitudes but more pulses are gen-

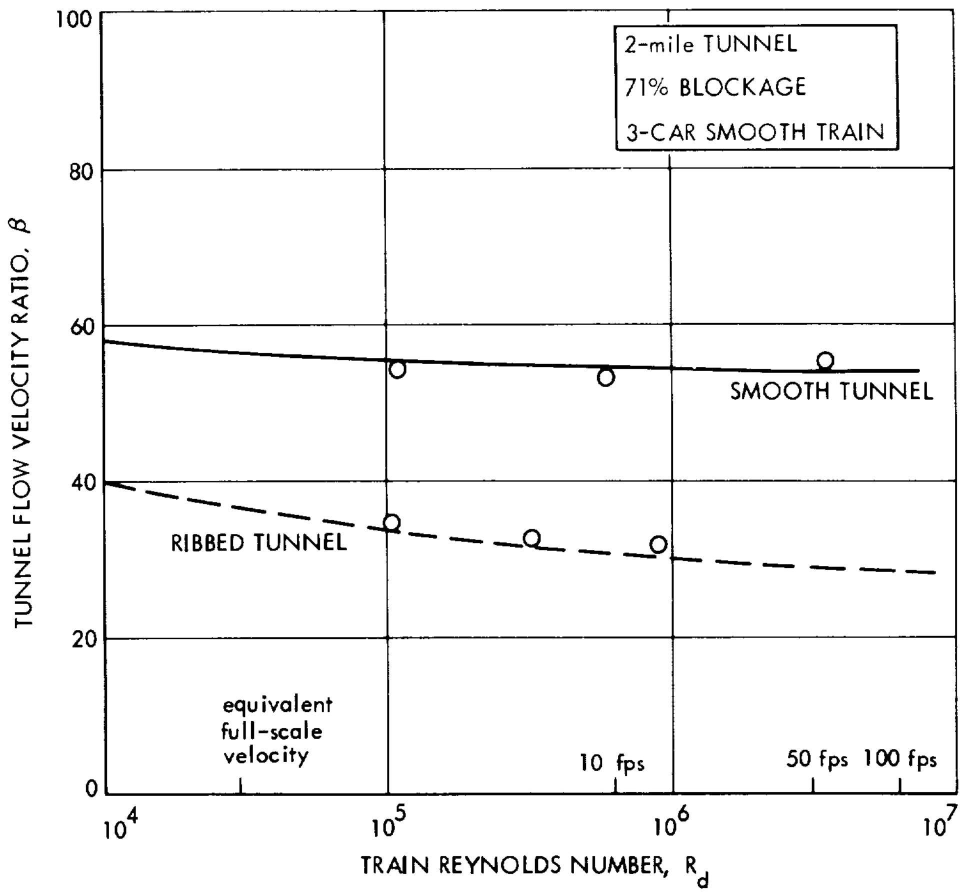


FIGURE 8. Ratio of Air Flow Velocity in Tunnel to Train Velocity as a Function of Reynolds Number

erated as the train hits relatively still columns of air as it passes each vent. Tunnel entrance shapes can be modified to further reduce the peak pressures.

UNSTEADY OPERATION

The time that an intra-urban subway train operates under steady-state conditions is only a portion of its total running time. Even though the train might reach its cruise speed in the order of a minute, the tunnel flow velocity far upstream or downstream may not stabilize for several minutes (if at all), depending upon the length of the tunnel. The forces on a train which must be overcome by the propulsion units may be broken into three categories: aerodynamic, inertial, and frictional.

The aerodynamic forces are a function of the velocity squared and the horsepower requirement is a function of the velocity cubed. Figure 9 presents the cruise horsepower required as a function of velocity for a three-car train. The rolling friction between steel wheel and rail is also shown in the figure for the

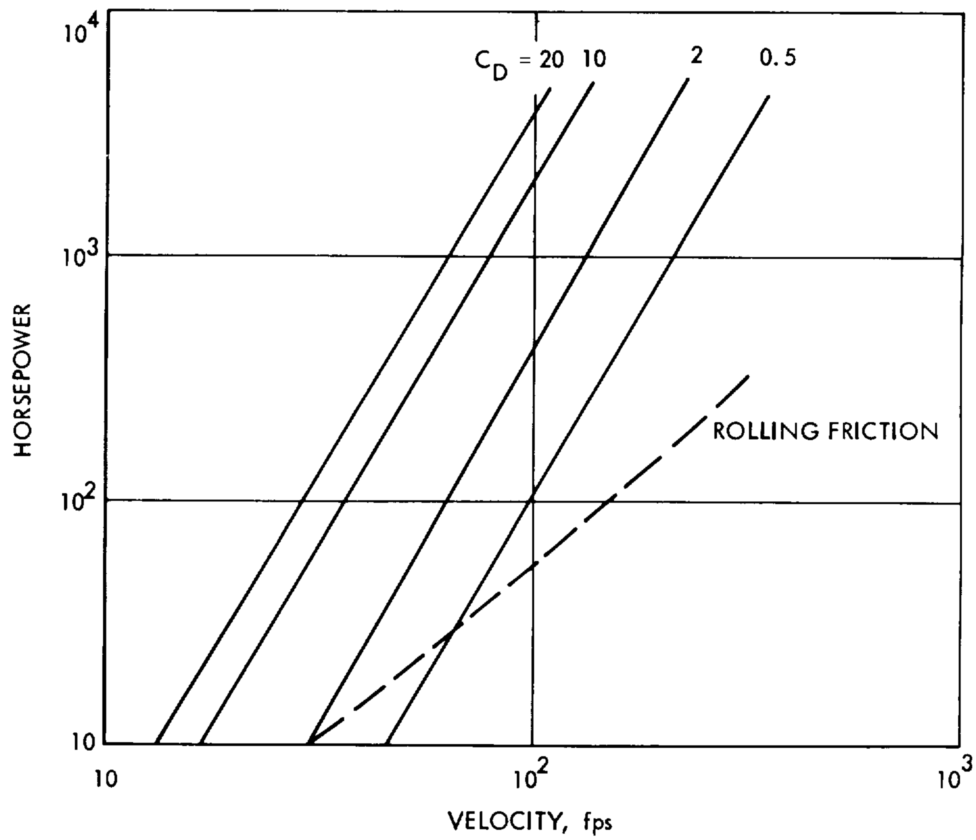


FIGURE 9. Aerodynamic and Rolling Friction Horsepower Requirement as a Function of Train Velocity

same twelve axle train. Under unsteady conditions, however, the flow in the tunnel as well as the train is accelerating. Therefore, additional forces are involved.

In order to best show the interrelationships between the power available and unsteady power requirements, an example case will be described. Passenger comfort requirements dictate that train accelerations do not exceed 0.1 g or about 3 ft/sec^2. The example, then, will consist of a 180,000 pound three-car 62% blockage ratio train (with realistic detailing and undercarriage) with a total of 1500 horsepower at the rail accelerating from rest at a vented station through a two-mile unvented tunnel. Cases are presented for both a smooth-walled tunnel and one with a typical internal ribbing configuration. The train moves slowly out of the station until it reaches an acceleration of 3 ft/sec^2 and holds that as long as possible with the power available. That is, this a constant power operation with a constraint of 3 ft/sec^2 as the maximum acceleration. Figure 10a presents the resulting velocity schedules in the two tunnels. It becomes readily apparent that the train is power limited in both cases with accelerations falling to less than 3 ft/sec^2 after only about 15 seconds. The train operating in the ribbed tunnel reaches its maximum cruise velocity of only 71 fps (48 mph) after about 80 sec. The train in the smooth tunnel would require almost 2½ minutes to reach its cruise velocity of just under 100 fps (68 mph). If there is another station stop at the end of the two mile tunnel and if the train were able to decelerate at 3 ft/sec² (this may not be practical due to the power re-

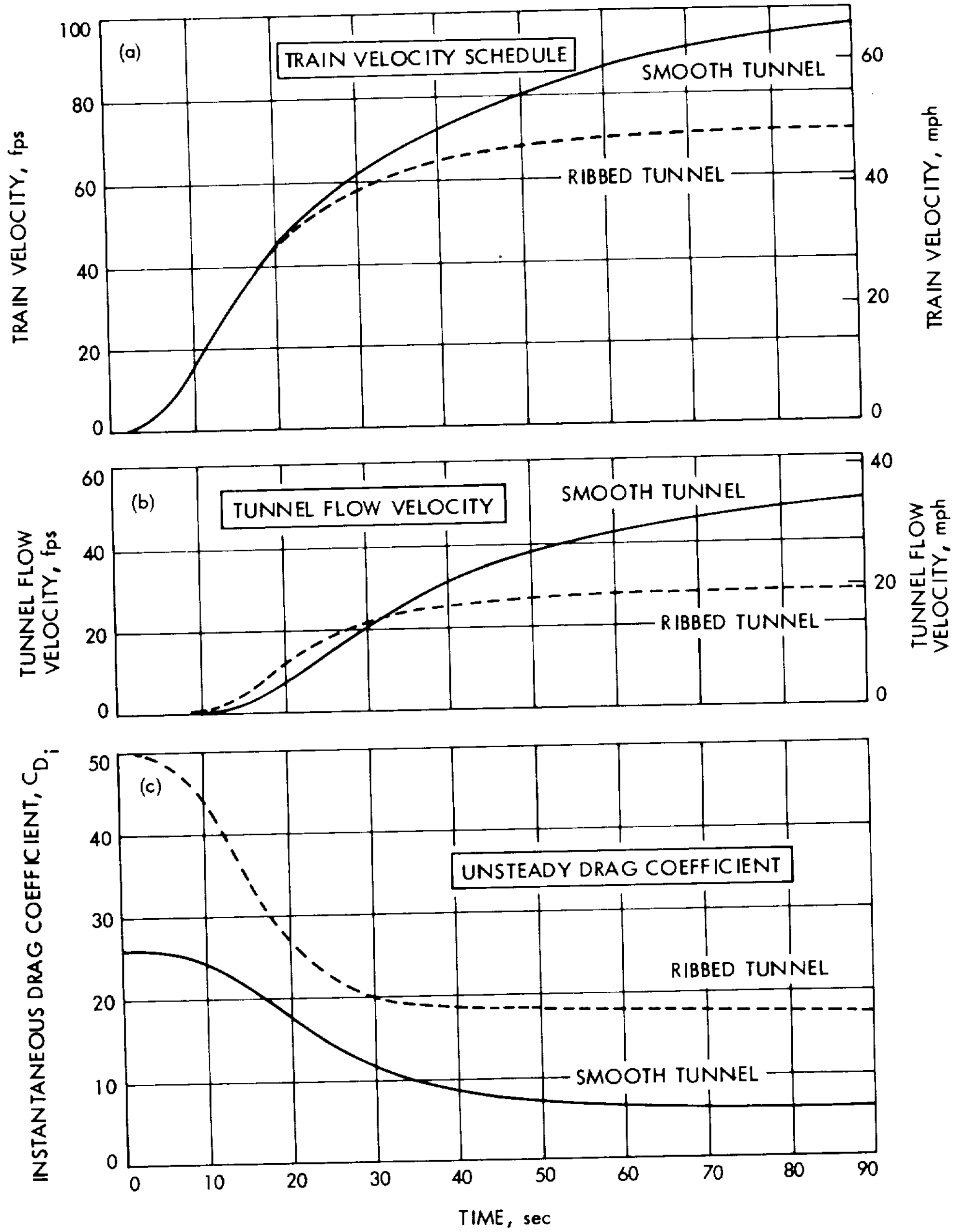


FIGURE 10. Unsteady Train Velocity Schedule, Resulting Air Flow Velocity in Tunnel and Instantaneous Train Drag Coefficient

quirements at cruise velocity and the inertia of the air flow in the tunnel as the train approaches a stop), the train in the smooth tunnel would have to start decelerating after only about 110 sec and would therefore never reach maximum cruise conditions. Using this schedule, then, the two mile station to station journey would take about 3 minutes in the ribbed tunnel and just over 2½ minutes in the smooth tunnel.

The air flow rate in the two tunnels during this operation is shown in Figure 10b. Notice that the flow accelerates much more rapidly in the ribbed tunnel even though it will equilibrate at a much lower velocity.

The instantaneous drag coefficient is shown in Figure 10c and reflects the situation exhibited by the tunnel flow velocity. That is, during the first few seconds the flow is essentially at rest and the tunnel appears to the train to be infinitely long; hence the drag coefficient is at its maximum. As the flow speeds up, the drag coefficient begins to decrease; and more rapidly in the ribbed tunnel since the flow acceleration is higher.

In order to gain perspective into the importance that aerodynamics plays in the system, horsepower requirements are shown in Figures 11 and 12. Figure 11

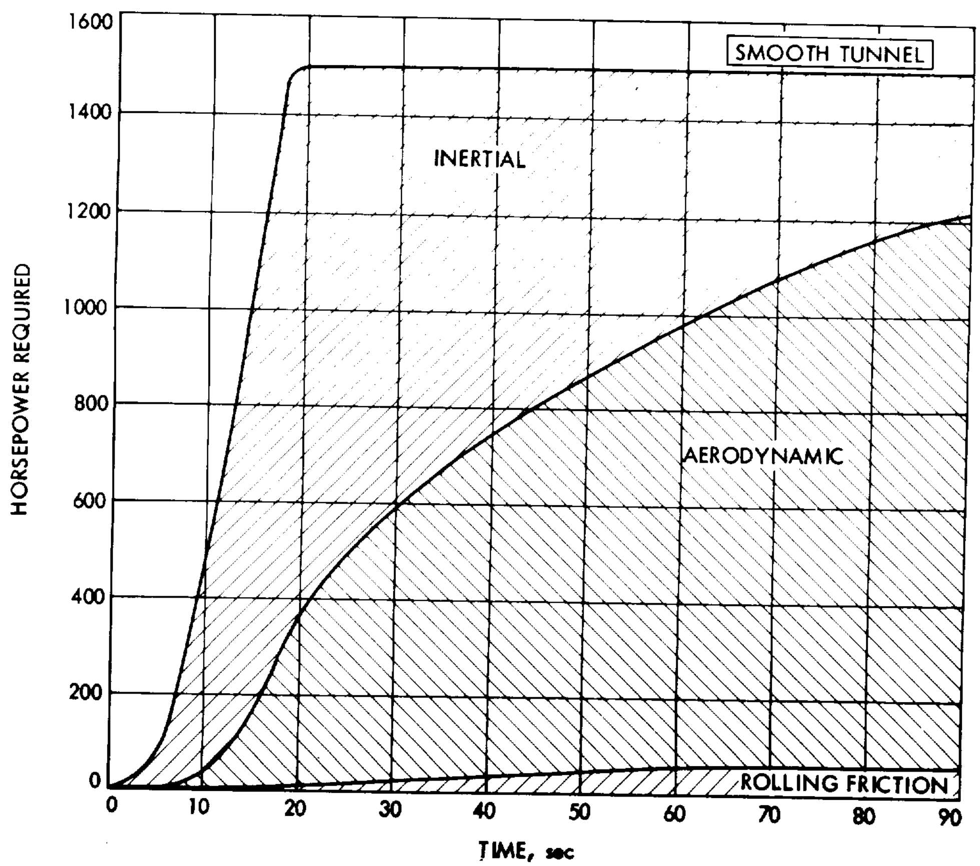


FIGURE 11. Component Parts of Constant Unsteady Horsepower Requirement for Train in a Smooth Tunnel

shows the relative importance of the three contributors that were considered in the system horsepower requirement for the smooth tunnel case. The rolling friction is only a weak function of the velocity and accounts for only about 60 horsepower at 100 fps. The horsepower required to accelerate the train mass is clearly dominant during the initial acceleration period. As the velocity is increased to say 50 fps (22 seconds) aerodynamic drag draws more than a fourth of the total system horsepower available and that swells to over 75 percent as 100 fps is approached. The situation is even more dramatic in the ribbed tunnel shown in Figure 12. Here the aerodynamic drag becomes the dominant component when the train is traveling only 40 fps and accounts for well over 90 percent of the available horsepower at cruise conditions which are limited to less than 50 mph.

HEAT REJECTION FROM TRAIN

As anyone who has ridden a subway knows, heat generation and discomfort to passengers on board and in stations is a major problem. Several authors have done some rather detailed analyses of the situation from a systems standpoint including passenger loading schedules (References 15 and 16). It has been found

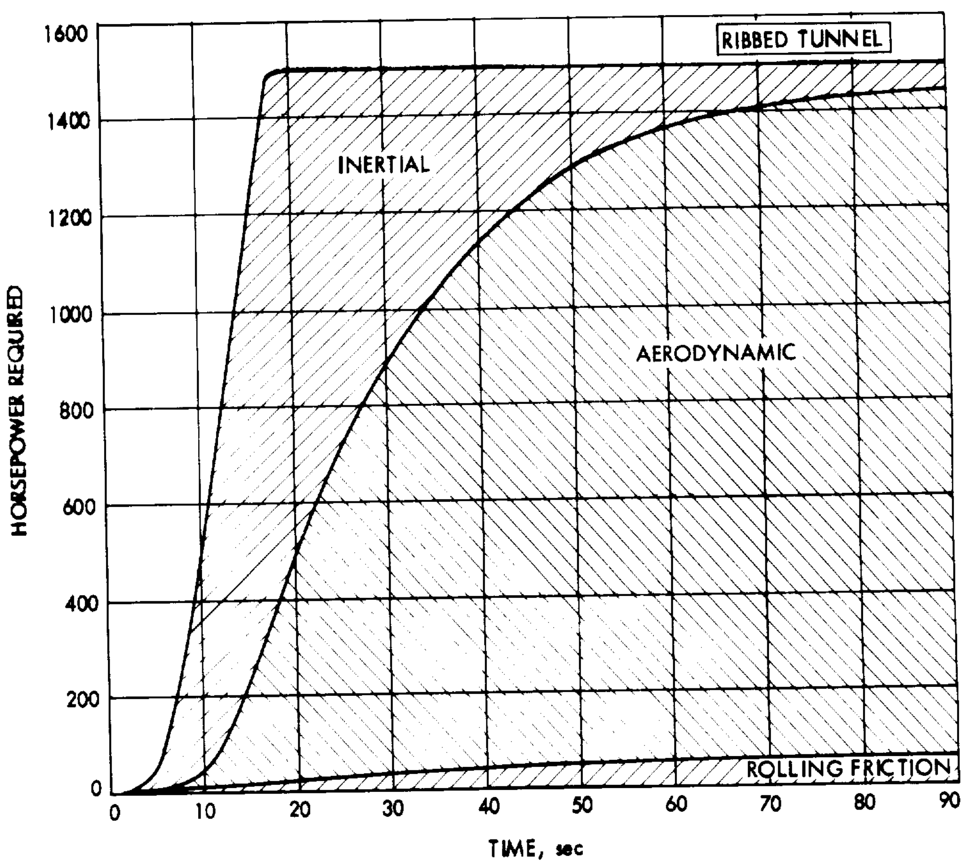


FIGURE 12. Component Parts of Constant Unsteady Horsepower Requirement for Train in a Ribbed Tunnel

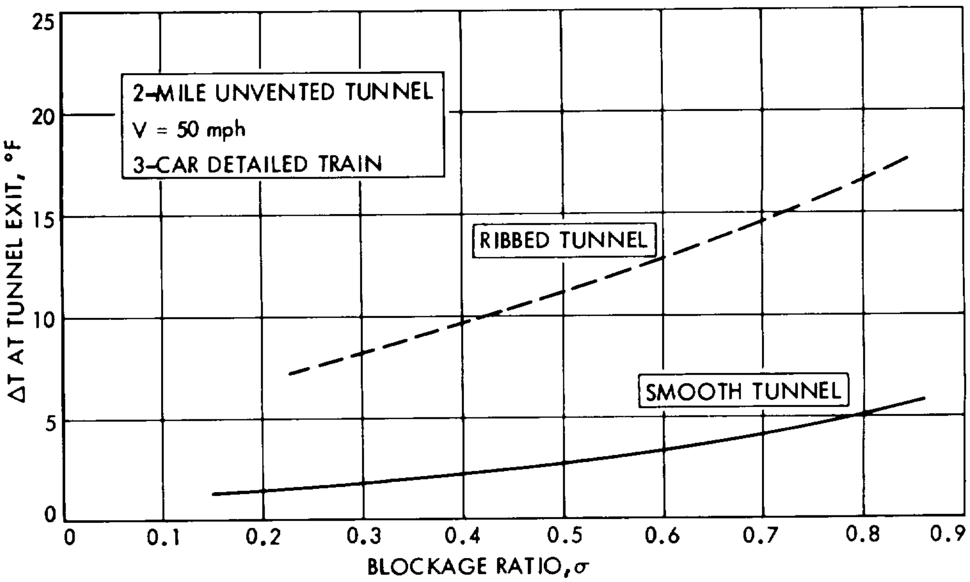


FIGURE 13. Temperature Rise at the Exit of a Tunnel During Continuous Train Operation

that necessary environmental controls in the subway portions can amount to 8 to 10 percent of the total system construction costs and operating power consumption may well be of the same order as that required for traction. Since the heat rejected from the train is clearly a function of the power expended, one readily concludes that higher blockage ratio systems may compound the problem. If the train is accelerating or pulling a grade, the situation is even worse.

Analyzing the problem from a very simplified approach can give one helpful insight to the magnitude of the problem. Figure 13 was constructed from such as analysis assuming steady-state operation, no heat transfer through the tunnel walls and the train surface at constant temperature. The formulation assumes that the power expended to move the train (against aerodynamic forces) during cruise is related to the flow velocity inside the tunnel which, if heat transfer through the tunnel wall is neglected, may be equated. That is the tunnel wall friction force integrated over the tunnel length is equal to the energy going into the air. The calculation was made for conditions of continuous operation; as soon as one train leaves the two mile tunnel another enters (not an unrealistic situation under rush hour conditions). The dramatic temperature rise in the ribbed tunnel is due to the fact that the flow velocity in the ribbed tunnel is lower and the train is in the tunnel longer.

SUMMARY

The purpose of this paper was to point out that high blockage subway systems have major aerodynamic considerations peculiar to this type of application. The aerodynamic forces may be one to two orders of magnitude greater than when traveling outside the tunnel. When an intra-urban train is traveling at cruise conditions, the aerodynamic forces can clearly dominate the power requirements.

When the train is operating under unsteady conditions, such as accelerating out of a station, the inertial forces are expected to be a dominant part of the power requirements only during the initial stages, even though the aerodynamic drag coefficient may be at its maximum. However, as shown in the examples presented in this paper, the aerodynamic forces are not only important under steady cruise conditions, but do severely limit the unsteady operation in terms of maximum power acceleration. It is obvious, then, that aerodynamics should be made an essential part of the system design trade-off study.

The aerodynamic data obtained in this experimental investigation, of which only a very small portion are included in this paper, cover a very extensive range of conditions which are thought to be applicable to actual subway-train rapid transit systems. Until these aerodynamic data are used in detailed system analysis, it will not be possible to determine if they cover a sufficient range of conditions to an adequate degree of precision.

APPENDIX

Basic Theoretical Model

A basic understanding of the aerodynamic characteristics of a tube-vehicle system can be obtained by the proper application of normal pipe flow laws along with momentum considerations. The formulation of a basic theoretical model is based upon these two principles. In order to further simplify the formulation, the following assumptions were made:

1. Constant vehicle velocity
2. Incompressible fluid
3. Single, unvented, constant diameter tunnel
4. Single axisymmetrical vehicle
5. Flow velocities are a function only of local cross-section area and are uniform

The friction factors used throughout the calculations are functions of Reynolds number and wall surface roughness, and have been determined from experi-

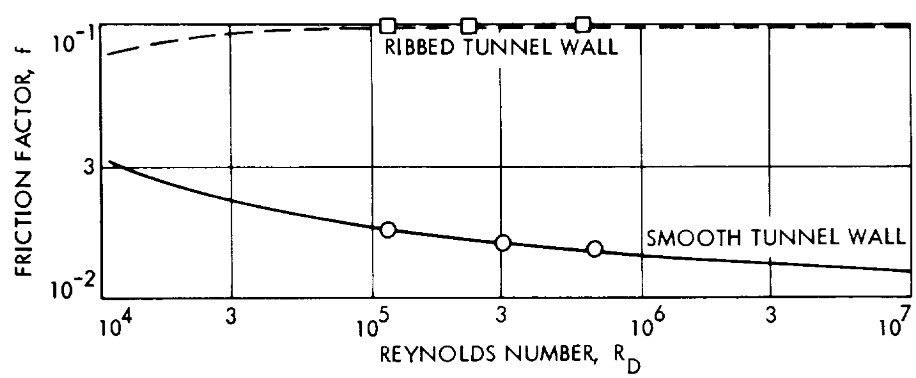
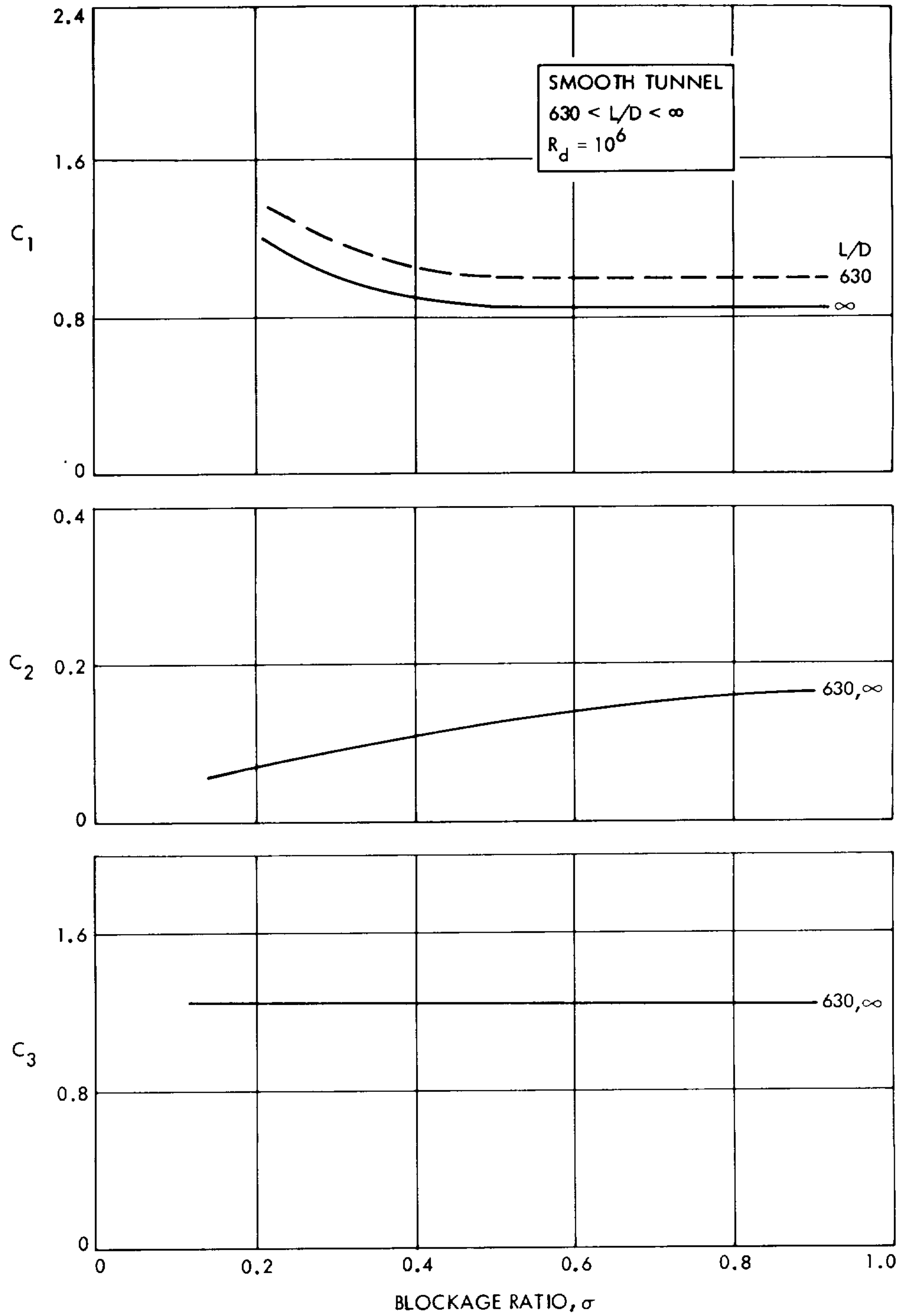


FIGURE A-1. Experimentally Determined Tunnel Wall Friction Factors Used in Analytical Model I

FIGURE A-2. Blockage Ratio Dependence of Momentum Coefficients



mental results. The friction factor on the tunnel wall away from the train is presented in Figure A-1 for both a smooth and ribbed tunnel. The friction factor in the annular region for the smooth train in a smooth tunnel is reasonably well represented for the examples given in this paper by the lower curve in Figure A-1. For the ribbed tunnel case, the annular friction factor is only somewhat larger (about 10-20 percent) than this lower curve when the train blockage is based upon the tunnel diameter inside the ribs. Due to separation, it does not come anywhere near approaching the upper curve in Figure A-1. Several momentum coefficients were included in the analytical model to handle the effects not considered in this simplified approach. The coefficient C_1 is the ratio of the experimentally measured pressure drop over the nose of the vehicle to that predicted from Bernoulli principles assuming uniform one-dimensional flow.

The coefficient C_2 takes care of the deviation in the pressure drop at the vehicle base from that assumed from simple annular pipe flow considerations.

The coefficient C_3 is the ratio of the experimentally measured pressure recovery in the wake immediately behind the vehicle to that calculated.

These experimentally determined coefficients are shown in Figure A-2 as functions of the blockage ratio. There are indications that at least C_1 is a function of the Reynolds number as well.

The vehicle drag is composed of four parts: (1) nose drag, (2) skin friction drag, (3) the pressure drop across the length of the vehicle due to the friction drag of the vehicle and the tube wall immediately adjacent to the vehicle, and (4) the base drag. Only the results of the theoretical analysis are included in the paper. The complete derivation can be found in Reference 13. This theoretical model is a logical extension to those developed in References 4 and 9. The inclusion of momentum coefficients and the effect of Reynolds number on the friction factors makes this formulation applicable over a wider range of conditions.

The drag coefficient may then be represented in component form as:

$$C_{D_{\text{total}}} = \left\{ C_{D_{\text{nose}}} + C_{D_{\text{skin friction}}} + C_{D_{\text{pressure drop}}} + C_{D_{\text{base}}} \right\}$$

where,

$$C_{D_{\text{nose}}} = \left\{ \sigma(2 - C_1) + 2(C_1 - 1) \right\} \left[\frac{1 - \beta}{1 - \sigma} \right]^2$$

$$C_{D_{\text{skin friction}}} = f_V \, \ell/d \left[\frac{1 - \beta}{1 - \sigma} \right]^2$$

$$C_{D_{\text{pressure drop}}} = \frac{\ell}{d} \frac{1}{(1 - \sigma)^3} \left[f_V \sigma (1 - \beta)^2 \pm f_T \, / \sigma (\sigma - \beta)^2 \right]$$

$$C_{D_{base}} = C_2 \left[\frac{1 - \beta}{1 - \sigma}\right]^2$$

In order to calculate the steady-state incompressible drag coefficient of a vehicle traveling in a simple tunnel, it is first necessary to determine β from the pressure equation.

The pressure difference between the two ends of the tunnel, which is equal to the pressure rise along the tunnel away from the vehicle less the pressure drop across the vehicle, has been experimentally found to be about one-and-three-quarters of a dynamic head. The pressure drop away from the vehicle is handled with well-known pipe friction equations. The pressure drop across the vehicle can be handled as four distinct regions: (1) pressure drop over the nose, (2) pressure gradient along the vehicle, (3) pressure drop at the vehicle base, (4) pressure recovery aft of the vehicle base. The formulation is as shown below:

$$\alpha^* = \left\{ f_U \frac{L_U}{D} - C_1 \frac{\sigma}{\beta^2} (2 - \sigma) \left(\frac{1 - \beta}{1 - \sigma}\right)^2 \right.$$

Tube end losses	Pressure gradient in tunnel upstream of train	Pressure drop over nose of train

$$- \frac{\ell}{d} \frac{1}{\beta^2 (1 - \sigma)^3} \left[f_V \sigma (1 - \beta)^2 \pm f_T \sqrt{\sigma} (\sigma - \beta)^2 \right]$$

pressure gradient in annular region along length of train

$$- \frac{C_2}{\beta^2} \left(\frac{1 - \beta}{1 - \sigma}\right)^2$$

pressure drop over base of train

* For normal open end case $\alpha = -1.75$.

$$+ 2C_\beta \frac{\sigma}{\beta^2} \left(\frac{1-\beta}{1-\sigma}\right)^2$$

Pressure recovery aft of train

$$+ f_D \frac{L_D}{D} \Bigg\}$$

Pressure gradient in tunnel downstream of train. f_D is somewhat larger than f_U depending on L/D and L_D/L_U, $(L = L_U + \ell + L_D)$.

where "+" is used when $\sigma > \beta$ and "−" is used when $\sigma < \beta$. f_U and f_D are the tunnel wall friction factors upstream and downstream of the train, respectively.

NOMENCLATURE

a	Train cross-sectional area, ft^2
A	Tunnel cross-sectional area, ft^2
C_D	Train drag coefficient, $\dfrac{\text{Drag}}{\frac{1}{2}\rho V^2 a}$
C_{D_∞}	Train drag coefficient in infinitely long tunnel
C_{D_i}	Unsteady instantaneous drag coefficient, $C_{D_\infty}(1-\beta)^2$
d	Train diameter, ft
D	Tunnel diameter, ft
f	Tunnel wall friction factor
f_T	Tunnel wall friction factor in the annular region
f_V	Train wall friction factor
R_d	Train Reynolds number, $\rho Vd/\mu$
R_D	Tunnel Reynolds number, $\rho UD/\mu$
U	Tunnel air flow velocity, ft/sec
V	Train velocity, ft/sec
α	Pressure difference between the two ends of the tunnel, $(P_{entrance} - P_{exit})/\frac{1}{2}\rho V^2$

β Ratio of tunnel flow velocity to train velocity, U/V

σ Blockage ratio, a/A

ρ Density, slugs/ft^3

μ Viscosity, slugs/ft-sec

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The Editor welcomes criticisms and constructive comments from the readers of this Journal.